Description of an Indoor Test Facility for Evaluating a Roof Integrated Cooling Concept

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Professional Paper

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ABSTRACT

An advanced roof test facility has been designed, constructed, debugged and made operational at the Florida Solar Energy Center. The facility is called the Diurnal Test Facility and was designed to provide high quality data which could be used for verification and improvement of the analytical model of the desiccant enhanced radiant cooling (DESRAD) concept. The effects of solar heating potential and nocturnal cooling potential are simulated in a controlled indoor environment and extensive measurements are made along and between the boundary surfaces. Air delivered to the test section is controlled to close tolerances in temperature, humidity and flow rate. Steady state conditions, step changes, functional changes or real weather conditions can be simulated. Accurate measurements are taken at the inlet and outlet of the test section to determine the amount of heat and mass transfer across the system. The facility is completely computer controlled. The control software, employing a self-tuning proportional-integral control methodology, was developed in house. A description of both the DESRAD concept and the Diurnal Test Facility is presented here along with examples of the model verification data and a brief measurement uncertainty analysis.

INTRODUCTION

The DESRAD cooling concept is a passive cooling approach which integrates a desiccant bed within a conventional roof structure to achieve both latent and sensible cooling in hot, humid climates. It's original development was by Fairey et al. (1986). Figure 1 illustrates the principle of operation which involves two modes -- a nighttime adsorption mode and a daytime desorption mode. The DESRAD concept uniquely takes advantage of nocturnal cooling potential to shed the heat of adsorption during the night, solar heating potential for desiccant regeneration during the day, and the moisture and thermal capacities of household materials to provide energy storage. Results from the analytical model by Swami et al. (1989) have shown significant cooling energy savings and peak electrical load reductions in hot, humid climates. A comparison of the cooling load requirements for a base house and a DESRAD house in Atlanta, GA is shown in Figure 2. The savings for total, sensible, and latent loads are 79%, 81%, and 75%, respectively.

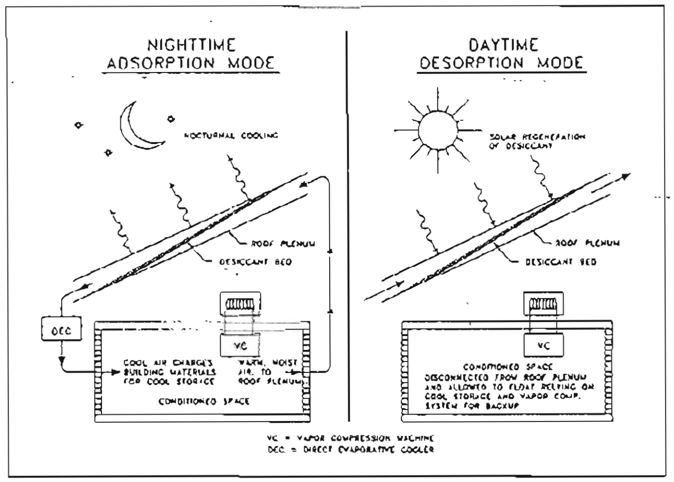


Fig. 1 DESRAD operating modes

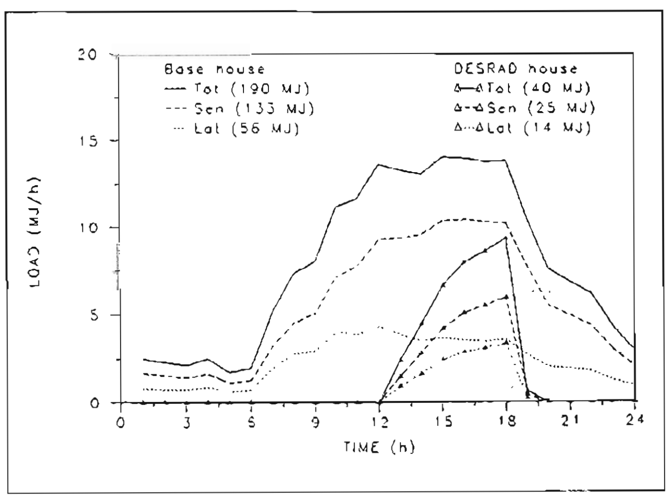


Fig. 2 Comparison of summer day cooling load requirements for base and DESRAD houses in Atlanta, Georgia

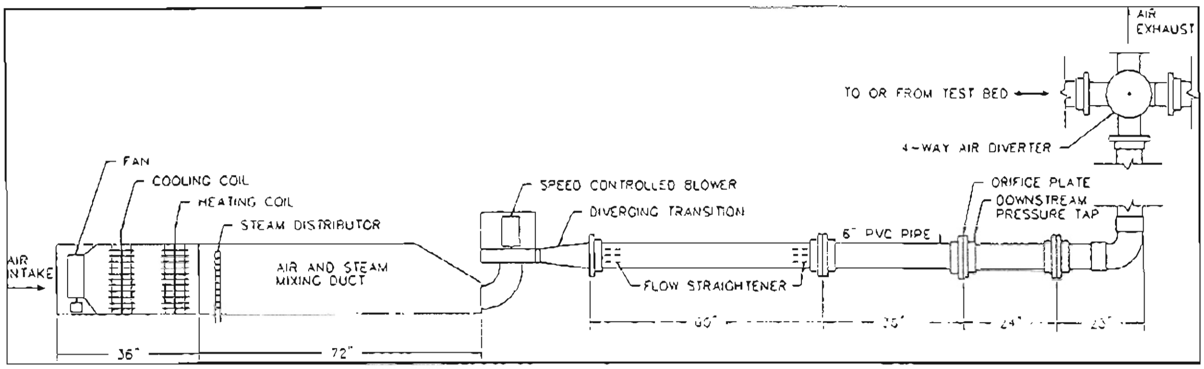


Fig. 3 Air handling apparatus layout

EXPERIMENTAL FACILITY DESCRIPTION

The Diurnal Test Facility (DTF) is being used to provide experimental data needed to verify the accuracy of the analytical model of the DESRAD concept. The DTF is capable of performing experiments requiring automatically varying air conditions, including temperature, humidity and air flow rate. This can be achieved using the air delivery apparatus of the test facility. Air flow can be controlled between 40 and 250 cubic feet per minute. The supply air dew point temperature can be controlled between 5.5°C (42°F) and 29.4°C (85°F). Supply air dry bulb temperature can be held between 4.4°C (40°F) and 71°C (160°F). The conditioned air is directed to a long rectangular duct with automatic temperature control on the top and bottom surfaces, and extensive instrumentation for temperature measurement along and between those surfaces. The top and bottom plate temperatures can be separately controlled between 4.4°C (40°F) and 76.6°C (170°F).

Air Delivery Apparatus

Figure 3 illustrates the basic layout of the air delivery apparatus. Description of the apparatus begins with a fan-coil unit. Three speeds are available to provide a base rate of air flow through the system. The coil is used to provide cooling and dehumidification using chilled water. Bolted to the fan-coil unit is a water-to-air booster coil used for heating.

Both the cooling and heating coils are flow controlled with electronic 3-way diverting valves. These valves are controlled through the data acquisition and control system with a 6 to 9 vdc input. The total flow through the two 62 watt (1/12 horsepower) circulating pumps is always constant and the valves simply divert the water proportionally to either the respective coil or back to the return line.

The next component is a steam distributor installed at the beginning of a 183 cm (6 foot) long duct to provide proper mixing of the steam and air. The steam distributor delivers steam to the air stream from a 9 kg/hr (20 lb/hr) steam humidifier. Another 3-way diverting valve proportionally diverts steam to either the distributor or to an external condensing coil and condensate waste system. Condensate from three locations, the cooling coil, the steam distributor, and the steam diverting valve, flows to a condensate pump which pumps to a gravity drain.

Following the 183 cm (6 foot) long mixing section is a direct-current blower connected to a variable-speed motor-controller. This system provides air flow control above the base air flow from the fan coil unit. The control feedback for the blower comes from a pressure transducer which measures the pressure differential across an orifice plate. The orifice plate was designed and constructed in house in accordance with the ASME Standard for Measurement of Fluid Flow, (ASME, 1985).

The final component of the air delivery apparatus is a 4-way air diverter. This air diverter allows the automatic redirection of air to either end of the test section duct, enabling simulation of bi-directional air flow with the minimum number of air flow measurements and dampers. Control of the air diverter is provided by an electronic modutrol motor which operates on a 4 to 7 vdc signal.

Chilled and Hot Water Storage

The chilled water storage system consists of a 2-ton chiller, with a water cooled condenser, plumbed to two insulated storage tanks each with their own circulation pump. The storage tank which services the top and bottom plate assemblies has a 151 liter (40 gallon) capacity. The tank which services the cooling coil in the air handling section has a 250 liter (66 gallon) capacity. Both circulation pumps are 30 watt (1/25 horsepower) and are used to continuously pump the tank water through the chiller. Temperature control of the chilled water is accomplished by adjusting the cut-in and cut-out pressures measured in the compressor suction line. Since the controlling adjustments are made at the chiller, both storage tanks are normally maintained at the same water temperature. However, it is possible to preferentially cool the chilled water tank serving the test bed plates through the control of an electronic 2-way proportional valve installed in the water supply line of the tank serving the cooling coil in the air delivery apparatus.

Hot water storage includes two 151 liter (40 gallon) insulated storage tanks. In order to keep the tanks well mixed, each tank has its own 30 watt (1/25 horsepower) circulating pump. One tank services the top and bottom plate assemblies and the other services the heating coil in the air handling section. Individual control of the hot water tank temperatures is accomplished through the use of optically isolated solid state relays. By controlling the temperature of the hot water tanks, it is possible to achieve much better control of both the air and plate temperatures. The solid state relays receive a control signal from the computer, thereby either blocking or passing the load current to the heater elements depending on the water temperature and the desired set point. The tank servicing the top and bottom plate assemblies has two 4500 watt heater elements, each in series with a 25 ampere solid state relay. The tank servicing the heating coil in the air handling section has two 2000 watt heater elements, each in series with a 10 ampere solid state relay.

Test Bed Apparatus

Primary experimental data are taken inside the test bed duct. Figure 4 is picture of the Diurnal Test Facility showing the test bed apparatus. At each end of the test bed, transition duct assemblies serve to change from 15 cm (6 inch) round pipe to 57 cm (22.5 inch) wide by 13 cm (5 inch) high aluminum duct. Each transition contains several custom fabricated components to assure uniform air flow into the test bed and to facilitate both dry bulb and dew point temperature measurements. Turning vanes move the air into a wider pattern, then a 64 mm (1/4 inch) thick perforated steel plate with 40% free area provides back pressure which tends to spread the air evenly across a vertical plane. Next is a tree of 64 mm (1/4 inch) O.D. copper tubing on which eight special limits of accuracy, Type T, shielded thermocouples are attached. These thermocouples are used to measure the average dry bulb temperature of the air entering the test bed. In addition to providing a mounting structure for the thermocouples, the copper tree is also drilled with eight small holes which allow an averaged sample of the test bed inlet air

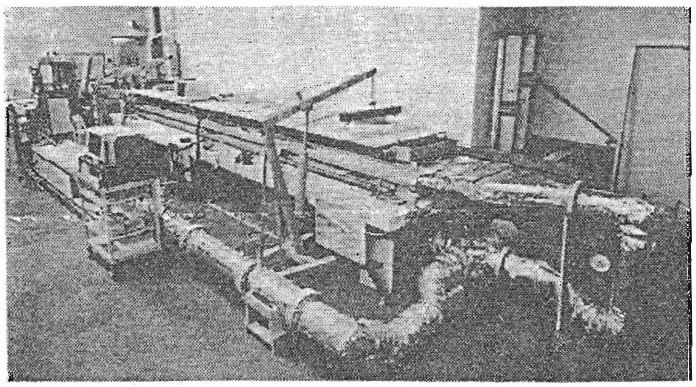


Fig. 4 Picture of the Diurnal Test Facility

to be drawn through the copper tubing to a remote chilled mirror dew point hygrometer. Air is drawn through the tubing and past the chilled mirror sensor by means of a vacuum pump. The airflow rate is maintained at 0.07 m^3/hr (2.5 ft^3/hr) by air flow meters. After the copper tree is another 64 mm (1/4 inch) thick perforated plate then a stainless steel honeycomb material that is used as a final flow straightener. The honeycomb dimensions were calculated according to AMCA Standard 210 for fan testing (AMCA, 1985), resulting in a cell cross section length of 1.14 cm (0.45 inch) and a cell length of 6.86 cm (2.7 inch).

The test bed duct apparatus consists of two major subassemblies - the top plate and bottom plate. Two aluminum channels form the sides of the duct. Each plate is identical except one faces up and the other down. The surfaces of the plates are made of four sheets of 32 mm (1/8 inch) thick tempered glass, each 61 cm (2 feet) wide by 152 cm (5 feet) long. Both sides of each glass sheet are instrumented with 12 special limits of accuracy type T thermocouples. The temperature of both the top and bottom plates can be separately controlled. Water is pumped, using two 373 watt (1/2 horsepower) pumps , through EPDM tubing mats which are in contact with the glass plates. Temperature control is provided by two electronic 3-way mixing valves. The valves mix water between the hot and chilled water supply lines to achieve the set point temperature for the particular plate which they are controlling. Feedback temperature sensing can be either from the thermocouples attached to the glass plate itself or from thermocouple probes installed in the inlet and outlet headers to the tubing mat. There are two supply and two return headers at each end of the test bed for each plate. By connecting every other tube in the tubing mat to the supply header and connecting the other tubes to the return header, a counter-flow design is achieved, giving a more even plate temperature. Behind the tubing mat is 12.7 cm (5 inches) of sprayed urethane foam encased in a wooden form.

Computer Control and Data Acquisition

The entire test facility is automatically controlled by computer. The control software is written in QuickBasic and runs on an 80386 based machine which serves as host to a computer front end data acquisition system. As shown in Figure 6, all input/output operations performed by the computer front end are initiated by instructions from the host computer. The computer front end performs all data measurement coming from thermocouples, RTD's (hygrometers), and pressure transducers and sends all data back to the host computer on fifteen second intervals where they are averaged for five minutes. The five minute averages are then sent via RS232 to a large computer for bulk data storage and data analysis. The computer front end also performs all analog outputs for control. All twelve of the control outputs are in the range of 0 to 10 vdc. They are calculated by the host computer program and sent from the computer front end every two seconds.

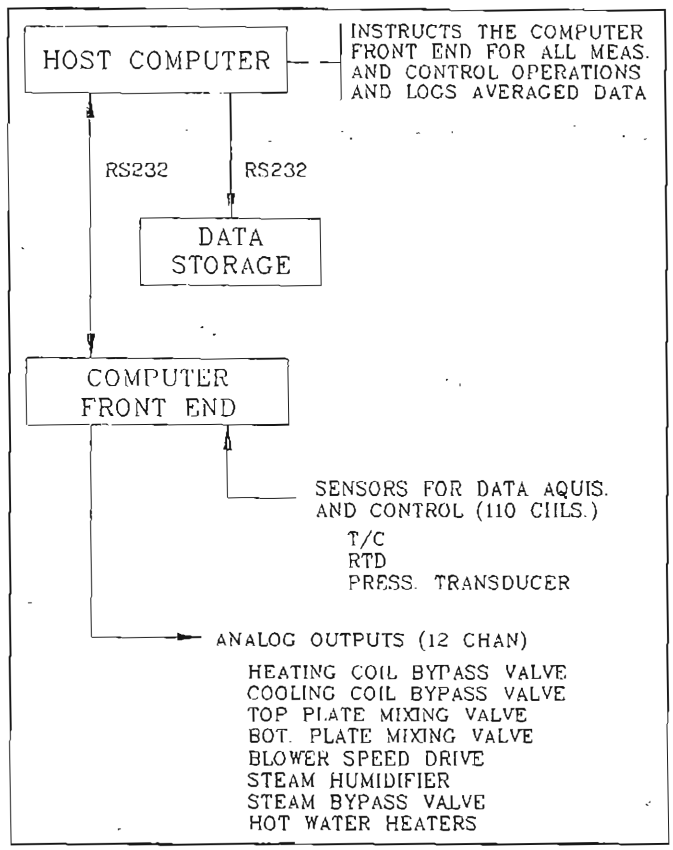


Fig. 5 Measurement and control hardware flow chart

The control program is a structured program working around a central driver. This structured format allows for the easy addition of new control routines and I/O instructions. All of the electronic valves and the blower speed controller are manipulated by proportional-integral (PI) control algorithms. These PI algorithms account for instantaneous errors as well as errors integrated over time to bring about stable control with little deviation from the desired set point. Derivative control is not used due to the slowly changing nature of the experiments currently being conducted.

One of the most important rules in proportional control is that the energy source and the energy sink be matched properly so that the controlled elements (in this case mostly electronic valves) can operate within their output span. For instance, if the water supplying a heating coil is very hot and the heating load is small then the control valve will remain closed or nearly closed all of the time. This condition does not allow for the settling action of proportional control, instead it is more like on/off control. For this reason, the control program constantly checks the output of the control elements and correspondingly adjusts some set points (i.e. hot water tank temperature, cold water tank temperature, and the steam humidifier output) to automatically fine tune the control capability. Additional self-tuning of the proportional and integral control constants, which affect the gain of the control element, is done to increase stability under varying load conditions.

MEASUREMENT UNCERTAINTY

An analysis of the measurement uncertainty for the test facility has been conducted according to the ANSI/ASME Standard PTC 19.1-1985, (ANSI, 1985). The standard provides a method for calculating and combining the random and systematic errors involved with calculating results from primary measurements. The systematic error is a fixed error and is represented by the bias limit. The random error is represented by the precision index and is based on the statistical variation (standard deviation) of the measured parameter. Nominal values of primary measurements were taken from a cross section of real experimental data.

TABLE 1
Input Parameters

Measured Parameter	Bias Limit	Precision Index	Nominal Values	Sensitivity Increment
Inlet dry bulb, °F	0.1	0.1079	77.79	0.2
Outlet dry bulb, °F	0.1	0.05247	84.08	0.2
Inlet dew point, °F	0.1	0.1637	67.99	0.3
Outlet dew point, °F	0.1	0.06927	59.34	0.3
Orifice dry bulb, °F	0.2	0.06545	78.05	0.2
Orifice dew point, °F	0.3	0.2076	68.01	0.3
Orifice ΔP, in. H_2O	0.01	0.000926	5.208	0.01
Barometric press, Pa	102	27.36	101300	102

The first step in the uncertainty analysis procedure is to develop a list of the input parameters as shown in Table 1. The measured parameters are any primary measurements which are used to calculate a result for which the uncertainty bounds are desired. Referring to Table 1, the bias limits were assigned based on sensor accuracy, sensor calibration, and data acquisition equipment accuracy. The precision index for each measured parameter was calculated using real data from the test facility. The sensitivity increment was chosen to be within one and two times the bias limit. A test to determine the importance of the size of the sensitivity increment showed that increments up to five times the bias limit made only a small difference in the result uncertainty.

A summary of the uncertainty of each final result is given in Table 2. Each result uncertainty is determined by propagating the precision index and bias limit of each measured parameter into the calculated result. Uncertainties are shown for the 95% coverage and the 99% coverage methods of calculation.

TABLE 2
Summary of Result Uncertainty

	q.sen	q.lat	air flow
Bias Limit	3.55E-03	7.38E-03	4.33E-05
Precision Index	3.01E-03	1.01E-02	4.97E-06
Uncertainty Percent (+/-)			
95% Coverage	4.4	4.9	0.1
99% Coverage	8.0	8.5	0.1

EXPERIMENTAL VERIFICATION OF THE MODEL

Experimental tests on a dual desiccant bed configuration have been completed using regular density silica gel. Actual test bed inlet conditions, shown in Figure 6, were passed as prescribed values to the analytical model and the outlet results were compared. Figure 6 shows how the dry-bulb temperature of the supply air was held constant while the dew point temperature was changed every eight hours to give a 37% step change in relative humidity. Experimental results gave a good cyclic moisture balance between the adsorption and desorption modes, moving about 0.9 kg (2 lb) of water with 35 kg (77 lb) of desiccant in 8 hours. For the given inlet conditions, the model's prediction of the central and outlet dew point and dry bulb temperatures are plotted against the experimental data in Figure 7.

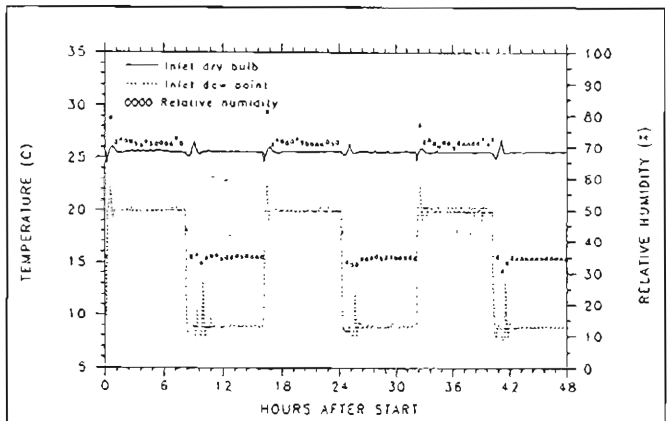
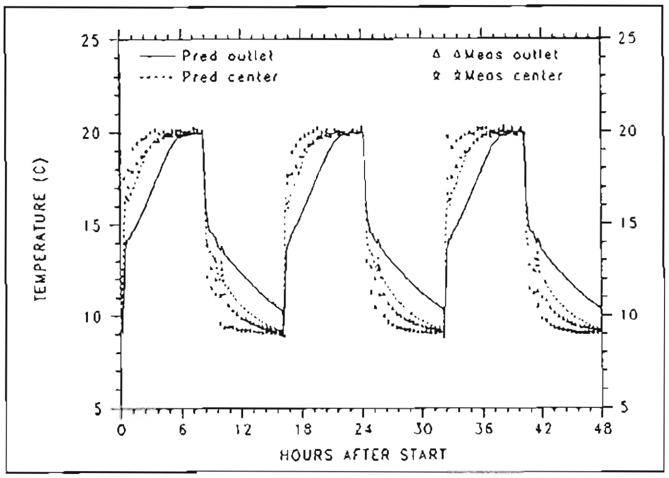


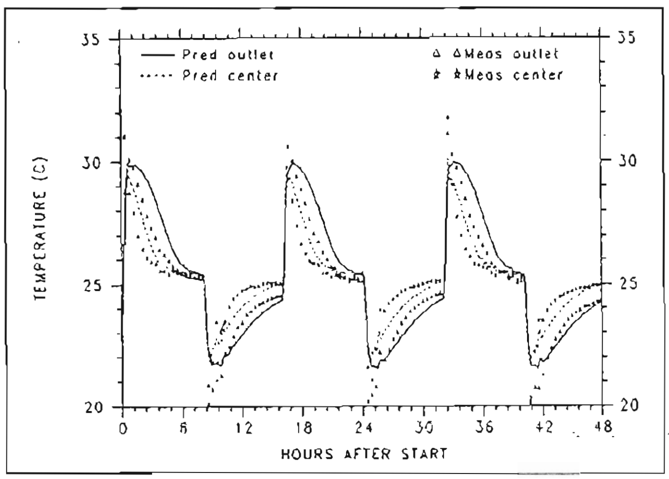
Fig. 6 Inlet air conditions for humidity cycling test of desiccant bed

The major problem that we have encountered thus far seems to be a partial utilization of the desiccant bed. The predicted versus measured data shown in Figure 7(1) represents a 65% reduction in bed utilization. When the model was re-run, with the amount of desiccant available for moisture transport reduced by 65%, the predicted results matched the experimental results well. The associated effect of this lowered bed utilization reduces the expected DESRAD energy savings by 15%. We are currently conducting more tests, both with the DTF and an environmental chamber, to isolate the source of this problem, which we suspect to be due to fluid dynamics, solid side resistance, and/or the desiccant isotherm.

Tests in the environmental chamber will investigate the problem from a material viewpoint. Several grades of silica gel will be tested at the same time, each with a different particle size, in order to investigate a possible problem with solid side resistance to moisture movement into and out of the desiccant particle. An investigation of the temperature dependence of moisture content as a function of relative humidity will also be made. Further testing with the DTF is designed to isolate possible problems with fluid dynamics, which may cause portions of the sloping desiccant beds to be bypassed.



(1)



(2)

Fig. 7 Comparison of predicted and measured temperatures for the two bed configuration
(1) dew point
(2) dry bulb

CONCLUSION

An experimental facility has been constructed to verify the accuracy of an analytical model of the desiccant enhanced radiant cooling concept. The results indicate that this building integrated system is a promising solar cooling alternative with potential for significant energy savings in hot, humid climates. However, before concrete recommendations can be made, more research is needed. Current experimental efforts show reduced bed utilizations of up to 65% compared to the model. Testing to isolate the source of the problem is underway using the Diurnal Test Facility and an environmental chamber.

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About this Report

This report was first presented at ASME International Solar Energy Conference, April, 1990 in Miami, Florida.

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